

ACOUSTIC TRANSFER VECTORS CONCEPT AND ACOUSTIC RADIATION ANALYSIS OF THE APPLICATIONS

KHÁI NIỆM VECTOR HÀM TRUYỀN VÀ ÁP DỤNG PHÂN TÍCH BỨC XẠ ÂM

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ABSTRACT

In the latest several decades, NVH (Noise-Vibration-Harshness) has become an important indicator for quality and comfort of cars. The structural acoustic performance of a car becomes an important issue in the design process. In car structures, the vibration of plates is not self-generated, but passed from the frame structure. Thus, the frame structure of a car is the important source of vibration and noise. The topography optimization, an optimization technique based on shape selection, is applied to problems of structural acoustic optimization. A case study on the application of topography on the frame structure is illustrated. The paper presents Acoustic Transfer Vectors and Modal Acoustic Transfer Vectors and their use in acoustic radiation prediction, particularly from the surfaces of frame structure. Acoustic Transfer Vectors are input-output relations between the normal structural velocity of the radiating surface and the sound pressure at a specific point in the field. The Modal counterpart gives a similar relation, but expressed in the modal coordinates of the radiating structure. The structural response, computed with a standard FE model such as NASTRAN, can be determined either directly in the frequency domain using a modal model, and this provides the boundary conditions for the acoustic radiation.

Keywords: Acoustic Transfer Vector, finite element method (FEM), boundary element method, structural acoustic radiation, topography optimization.

TÓM TẮT

Trong những năm gần đây, NVH (Tiếng ồn - rung động - xóc) là một chỉ số quan trọng cho chất lượng và sự thoải mái trên ô tô. Trong quá trình thiết kế, một vấn đề quan trọng là ảnh hưởng âm thanh của cấu trúc một chiếc xe. Trong cấu trúc xe, rung động của tấm không phải là tự tạo ra, nhưng được truyền từ cấu trúc khung. Do đó, cấu trúc khung của một chiếc xe đóng vai trò như một nguồn rung động và phát ra tiếng ồn quan trọng. Tối ưu hóa hình thái học, một kỹ thuật tối ưu hóa dựa trên sự lựa chọn hình dạng, được áp dụng để giải quyết vấn đề về tối ưu hóa cấu trúc. Tối ưu hóa hình thái học cấu trúc dạng khung là một áp dụng trong nghiên cứu này. Bài báo trình bày các vector truyền âm và các phương thức vector truyền âm được sử dụng chúng trong dự báo bức xạ âm, đặc biệt các bức xạ âm phát ra từ các bề mặt của cấu trúc dạng khung. Vector truyền âm là một thông số vào và ra giữa vận tốc pháp tuyến kết cấu của bề mặt bức xạ và áp suất âm thanh tại một điểm cụ thể trong trường âm. Các phương thức mô phỏng cho một mối quan hệ tương tự, nhưng thể hiện trong tọa độ phương thức của cấu trúc bị bức xạ. Thông số của đáp ứng cấu trúc được tính bằng mô hình FE chuẩn như NASTRAN, có thể xác định được trực tiếp trong miền tần số với việc sử dụng mô hình phương thức và điều này là điều kiện biên cho bức xạ âm.

Từ khóa: Vector truyền âm; Phương pháp phần tử hữu hạn (FEM); Phương pháp phần tử biên (BEM); Bức xạ âm-cấu trúc; Tối ưu hình thái học.

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1. INTRODUCTION

In the automotive and other industries where there is frame structure, which radiate noise, reducing and controlling that noise is often an important functional performance criterion of the design. The purpose of this paper is to show analysis of the folded plate structural acoustic radiation pressure. Many numerical methods, such as the finite element method (FEM) [3], the boundary element method (BEM) [4], the statistical energy analysis (SEA) [3,4] and the energy flow analysis (EFA) [6], have been developed to simulate the structural acoustic performance of a car. Different methods must be used based on the design objective.

Two boundary element formulations can be used to derive the Acoustic Transfer Vectors (ATV): the direct formulation in terms of pressure and velocity as boundary variables and the indirect formulation in terms of single and double layer potentials (velocity jump and pressure jump respectively). The direct formulation is the most straightforward approach but the indirect formulation is the most general. Indeed, when both sides of the boundary radiate, the direct formulation fails to give a correct representation. Because it is expressed in terms of velocity jump and pressure jump, the indirect formulation is well suited to such problems. Both formulations (using a collocation technique for the direct formulation and a variation approach for the indirect formulation) will be derived in the first section. Finally, and for sake of simplicity, the absorbent boundaries will be presented in the mathematical derivations. The reader should be aware that, although not presented, these types of boundary conditions or special treatments are possible in inverse boundary element methods.

The process presented in this paper, using Acoustic Transfer Vectors, provides a dramatic reduction in the overall computation time for acoustic radiation prediction in such applications, without losing the accuracy inherent in a proper solution of the acoustic wave behavior around the radiating surface using BEM.

2. ACOUSTIC TRANSFER VECTOR BASED ACOUSTIC RADIATION PRESSURE

The acoustic transfer vectors from the radiating surface to specified field points are evaluated in the first step across the frequency range of interest at fixed frequency intervals. In the second step, the acoustic response $p(\omega)$ in the field points is calculated for all loading conditions by combining the ATV with the normal structural velocity boundary condition vector at any frequency within the range. This ATV-response calculation is a vector-vector product, given as,

$$p(\omega) = \{atv(\omega)\}^T \{v_n(\omega)\} \tag{1}$$

Where $\{atv(\omega)\}$ is the acoustic transfer vector; ω is the angular frequency; $\{v_n(\omega)\}$ is normal velocity on the surface.

An important observation is the fact that in case of sound wave propagation in an open space the fluid domain around the radiating object exhibits hardly any resonant behavior. Therefore, ATV's are rather smooth functions of the frequency, and coefficients can be accurately evaluated at any intermediate frequency, called slave frequency, using a mathematical interpolation scheme based on a discrete number of frequencies, called master frequencies. It is important to note that the structural normal velocities cannot be similarly interpolated, since these directly depend upon the highly resonant dynamic behavior of the structure.

Another important advantage is that these frequency dependent ATV's can also be used for contribution analysis, i.e. by a 'partial' vector-vector product only taking into account the normal velocity boundary conditions on part of the radiating surface, i.e.

$$p_c(\omega) = \sum_{e=1}^{n_p} \{atv^e(\omega)\}^T \{v_n^e(\omega)\} \tag{2}$$

where by the superscript e denotes an element contribution. This way, the contribution of groups of elements, corresponding to distinct panels of the structure, can be derived, providing more insight into the noise generation mechanisms.

The engineering process to compute the structural normal velocity on a vibrating surface relies usually on the structural finite element method, and often on the modal superposition approach, where the structural response is expressed as a linear combination of the mode shapes of the body, as in the following relation:

$$\{v_b(\omega)\} = i\omega[\phi_n]\{mrsp(\omega)\} \tag{3}$$

where i is the i^{th} mode shape; $[\phi_n]$ is the matrix composed of the modal vectors, projected on the local normal direction of the boundary surface, and $\{mrsp(\omega)\}$ is the modal response (vector of the modal participation factors) of the structural model at a given excitation frequency.

Combining Eq. (3) with Eq. (1) leads to,

$$p(\omega) = i\omega\{atv(\omega)\}^T [\phi_n]\{mrsp(\omega)\} \tag{4}$$

$$\text{where } i\omega\{atv(\omega)\}^T [\phi_n] = \{\matv(\omega)\}^T \tag{5}$$

is called the Modal Acoustic Transfer Vector $\{\matv(\omega)\}$, which can be directly combined with the modal response vector to give the sound pressure at a field point:

$$p(\omega) = \{\matv(\omega)\}^T \{mrsp(\omega)\} \tag{6}$$

MATV's are the modal counter part of the ATV's. They express the acoustic transfer from the radiating structure to a field point in modal coordinates and therefore contain the acoustic contributions from each individual structural mode. The acoustic response in the field point is obtained by recombination of the MATV with the corresponding structural modal responses. Working in modal coordinates results in an important data reduction. It's clear however that MATV's are no longer independent from the structural model as they are linked to the structural modal basis. Whenever the structural modal basis changes, e.g. due to structural design modifications, the set of MATV's needs to be reevaluated. From Eq. (5) it is clear however that for a given structural modes set, the corresponding set of MATV's can easily be re-generated by projecting the ATV's, independent of the structural model, into the modal space. It's important to note that this quick generation of MATV's by projecting the ATV's into a new modal basis is only valid if the acoustic configuration has not been changed due to the structural design modification.

As a result, the frequency dependent acoustic pressure is obtained, and can be represented in the form of waterfall diagrams or as a color map on the field point mesh.

3. THE EXAMPLE OF STRUCTURE MODEL AND NUMERICAL RESULTS

+ The finite element model

The folded plate is a symmetry structure with crank angle α as shown in figure 5.1. The length of the folded plate is 0.3 m, the width is 0.2 m and the thickness is 0.001 m. The density of this folded plate structure is 7800 kg/m³ and Poisson's ratio of the folded plate structure is 0.3. The acoustic velocity of the air is 343 m/s and the air density is 1.21 kg/m³. After optimization, the above structural model is changed from a plate to a shell [1, 2].

The plate is divided into the four-node quadrilateral shell elements. The finite element model of the folded plate consists of 2400 elements and 2500 nodes. The analysis frequency range is 15 to 300 Hz. That is the acoustic frequency band mainly exists inside the car. All edges of the plate are clamped (200 nodes).

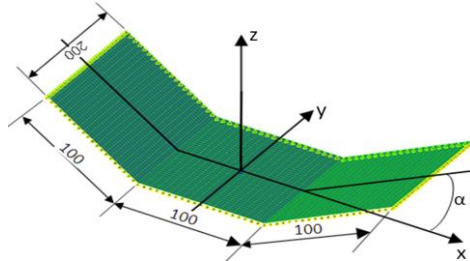


Figure 1. The finite element model of the folded plate structure

It is possible to compare the sound pressure results for the folded plate model by computing the noise transfer function of the structure folded before and after optimization.

+ Topography optimization of structures [1]

Usually, the natural frequency of the first mode has the largest contribution to the dynamical characteristics of a plate structure. The first-order mode of vibration is the one of primary interest. Maximizing the natural frequency of the first mode shape will also increase the natural frequency of higher modes and the stiffness of a structure. In this study, the natural frequency of the first mode shape of a folded plate structure is taken as the objective function. By introducing beads or swages to the bracket, the natural frequency of the first mode shape of a folded plate structure is maximized. The optimization software Altair OptiStruct is used to optimize the design of folded plate structures. The shape of folded plate structures is defined by the finite element nodes. The nodes in the design region are taken as the design variables. The finite element mesh is generated by MSC/NASTRAN automatically. The mesh generation parameters are input by the designer. The selection of the nodes to move is automatic. As shown in Figure 2, the new position of the nodes and elements are determined through the parameters selected by the designer as minimum width (m), folding angles (α), draw angle (β), draw height (h) and manufacturing constraints. The process of topography optimization is to move nodes following the direction of the normal vector of an element. The topography optimization had been done iteratively and automatically until the convergence conditions were met. After optimizing, the results have been shown in Figure 3

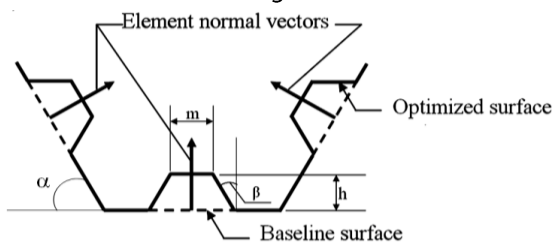


Figure 2. Beads created using the element normal vectors

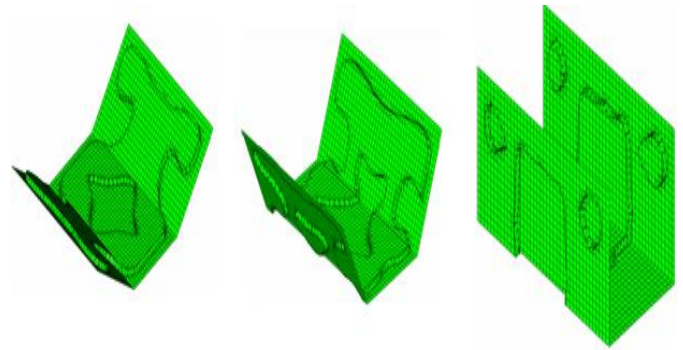


Figure 3. Topography model of the folded plate after optimization with folding angles $\alpha = 30^\circ, 60^\circ$ and 90°

The acoustic transfer function represents the contribution, at the angular frequency ω , of the structural modes to the sound pressure and its calculation proceeds through the following steps:

(1) The calculation of the acoustic transfer matrix:

- + Calculation by NASTRAN (SOL 103) of the structural model eigenvectors;
- + Transfer of such eigenvectors to the acoustic model. In addition, to proceed with the calculation of the sound pressure vector it is needed;
- + To calculate by NASTRAN (SOL 111) the vector of modal coordinates;
- + To apply on it the modal acoustic transfer matrix.

(2) The calculation of the acoustic transfer vector:

- + ATVs are evaluated within a frequency range from 1 to 300 Hz, with steps of 10 Hz;
- + A cubic spline interpolation scheme is used to estimate the ATV at all excitation frequencies in the range of interest. The requirements on the number of master frequencies, from which the ATV curves are obtained at any frequency, depends on the shape and smoothness of the frequency dependency of the ATV [7].

(3) Having chosen the SYSNOISE MATV procedure, four sequential analyses are needed:

- + Generation of the file *.bdf by the software Hyper Mesh;
- + ATV computation by SYSNOISE [10];
- + Acquisition of the NASTRAN output, eigenvectors and modal coordinates, and conversion of the acoustic transfer matrix;
- + MATV solves, to obtain the pressure levels by multiplying the modal acoustic transfer matrix and the vector of modal coordinates.

The acoustic pressures level in all calculations before and after optimization are shown in Figure 4~6. The acoustic pressure results computed using the acoustic transfer function with folding angles $\alpha = 30^\circ, 60^\circ$ and 90° have lower acoustic pressure after optimization compared with the models before optimization. The optimized structure reduces the sound pressure at the driver's ear.

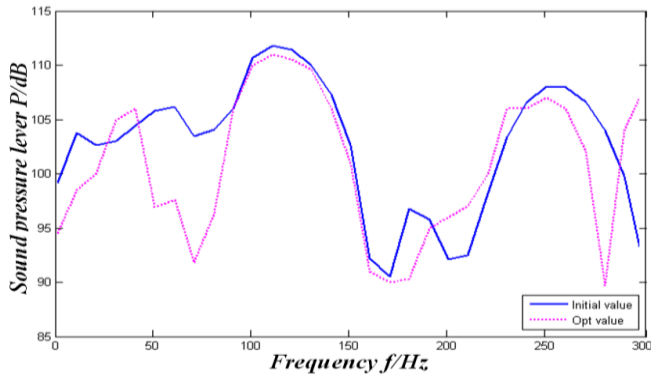


Figure 4. Acoustic pressure before and after optimization for folding angle $\alpha = 30^\circ$

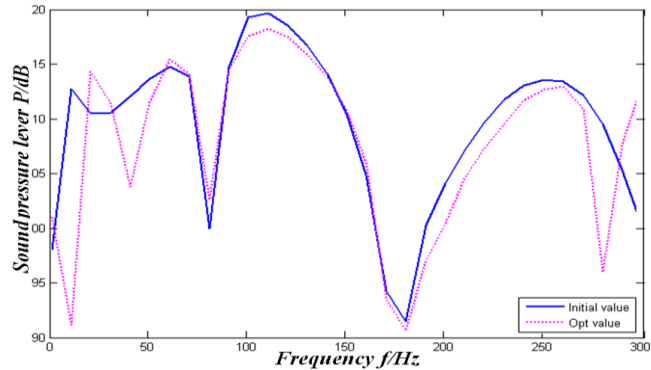


Figure 5. Acoustic pressure before and after optimization for folding angle $\alpha = 60^\circ$

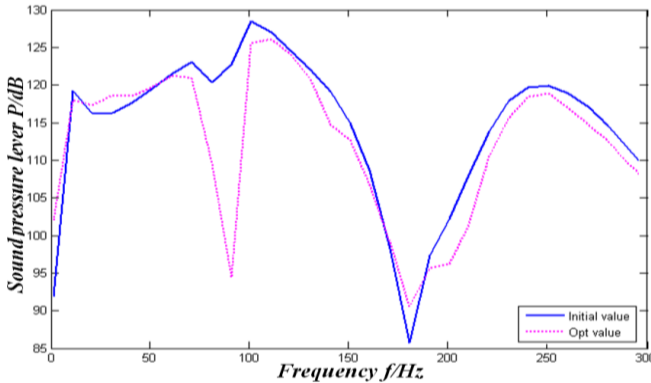


Figure 6. Acoustic pressure before and after optimization for folding angle $\alpha = 90^\circ$

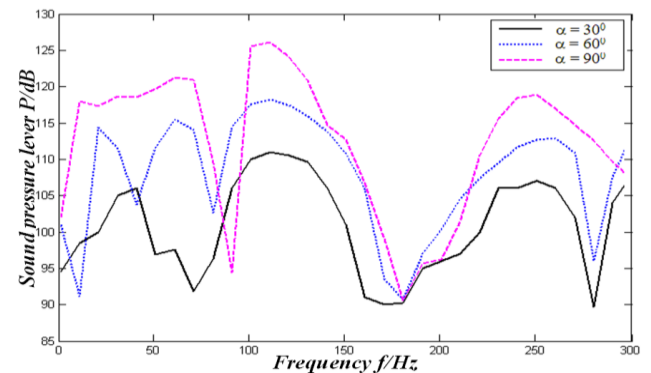


Figure 7. The acoustic pressure of optimized folded plates under different folding angles α

It can be seen from figure 7 that the structural acoustic pressure of the optimized plates is different when α has different values. This difference helps the designer to select the optimal folding angle α in the early stage of design. The above results of acoustic transfer function analysis have shown that the structural acoustic pressure of folded plate structures can be reduced by using topography optimization method.

4. CONCLUSION

This paper aims at presenting a methodology to calculate a noise transfer function and create an objective function used as a valuation analysis of acoustic radiation of the structure before and after topography optimization [11].

A new procedure for the efficient numerical calculation of sound radiation problems has been developed. The software implementation of the methods is done side by side with the method development. Problems related to the acoustic-structure interaction between materials and acoustic waves are analyzed and solved by computer simulations, which are efficient tools in the testing and optimizing of model parameters. The design process can be improved and the development cycle shortened with the above analysis. The structure topography optimization was to reduce the noise transfer function as well as of the acoustic pressure may be considered as a significant improvement.

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